TORQUE-LIMITED ELECTRIC SERVO SYSTEM FOR DEPLOYING A VEHICLE SNOW CHAIN TRACTION SYSTEM

BACKGROUND OF THE INVENTION

Field of the Invention

This invention relates to vehicle ice and chain traction systems which may be both rapidly deployed and rapidly retracted. More particularly, it relates to electrically-powered apparatuses for deploying a mounted system so that the components thereof are moved from a stowed configuration to an operable configuration.

Description of Related Art

Rapidly-deployable chain traction systems, which may be characterized generally as systems which fling short chain or cable segments beneath a road tire, have been known for some 90 years. Such a system is disclosed in U.S. Pat. No. 1,045,609 and in German Pat. No. No. 266,487 to W. H. Putnam for an ANTISKIDDING DEVICE. Throughout the years, various modifications and improvements have been made by numerous inventors. The following list is a representative list of a dozen other U.S. patents issued in this field:

US Pat. No. 1,150,148 for a TRACTION AND ANTISKIDDING DEVICE;

US Pat. No. 1,223,070 for an ANTISKIDDING DEVICE FOR VEHICLES;

US Pat. No. 1,374,252 for an ANTISKID DEVICE FOR AUTOMOBILES;

US Pat. No. 1,381,001 for a NON-SKID DEVICE FOR MOTOR AND OTHER VEHICLES;

US Pat. No. 1,975,325 for an ANTISKID CHAIN AND MEANS FOR APPLYING AND REMOVING SAME;

US Pat. No. 2,241,923 for an AUTOMATIC EMERGENCY TRACTION DEVICE FOR AUTOMOBILES;

US Pat. No. 2,264,466 for an ANTISKID DEVICE FOR VEHICLES;

US Pat. No. 2,277,036 for an ANTISKID DEVICE;

US Pat. No. 2,283,948 for an AUTOMOBILE TRACTION DEVICE;

US Pat. No. 2,442,322 for an ANTISKID DEVICE;

US Pat. No. 4,299,310 for an ANTISKID DEVICE FOR MOTOR VEHICLES;

US Pat. No. 4,800,992 for an ANTI-SKID DEVICE; and

US Pat. No. Des. 286,524 for ANTI SKID CHAIN UNIT FOR VEHICLE TIRES.

Referring now to the prior-art system of Figure 1, a modern rapidly-deployable chain traction system 100 is depicted in its deployed configuration in this rear elevational view drawing. The chain traction system 100 is removably affixed to a drive axle 101 which incorporates a differential unit 102. Inner and outer road wheels (103A and 103B, respectively) are mounted on the visible half of the drive axle 101. On each road wheel (103A and 103B) is mounted a rubber tire (104A and 104B, respectively). The chain system 100 includes a friction drive disc 105 to which a plurality of chain segments 106A, 106B and 106C are attached. Chain segment 106A is depicted as being below the road surface 114, which is normally covered with a layer of snow or ice when the chain system 100 is in the deployed configuration. The friction drive disc 105 is rotatably mounted on a spindle 107 which is affixed to a support member 108 which is pivotally mounted to a mounting bracket 109. The mounting bracket is, in turn, bolted to the U-bolt shackles 113 which secure the suspension leaf springs 112 to the drive axle 101. The chain system 100 also includes a pneumatic cylinder 110 that is bolted to the mounting bracket 109. The pneumatic cylinder 110 has a slidable piston 111 that is held in a normally retracted position within cylinder 110 by spring biasing when pressure within cylinder 108 equals ambient pressure. The outer end of piston 111 is connected to support member 108. In the deployed configuration, the outer rim of friction drive disc 105 is pressed against the sidewall of tire 104A by a biasing force applied to support member 108 by piston 111. The biasing force is provided by pneumatic pressure inside pneumatic cylinder 110 which overcomes the spring biasing and causes piston 111 to extend. As the tire 104A rotates, the friction drive disc 105 also rotates with the chain segments 106 extended more or less radially therefrom. Thus each chain segment 106 is flung, sequentially, beneath the tread portion of tire 104A. In order to retract the system and disengage the friction drive disc 105 from contact with the sidewall of tire 104A, pneumatic pressure to pneumatic cylinder 110 is cut off, causing piston 111 to retract within cylinder 110 and raising the support member 108, the rotatably attached friction drive disc 105

and the attached chain segments 106. In the retracted configuration, the chain segments 106 do not touch the road surface 114.

Referring now to the side view of the modern prior-art modern rapidly-deployable chain traction system 100 of Figure 2, the pneumatic deployment components and the mounting system are shown in greater detail. The mounting system shown is designed for use on vehicles which have a beam or live axle (i.e., one which incorporates a differential) 101. On each side of the vehicle, the apparatus mounts to U-bolt shackles which commonly secure the axle to a set of leaf springs 112 or an air bag assembly (not shown). If no U-bolts are present on the axle, a new set of U-bolts may be installed thereon and used to secure the system. In either case, the new mounting system is designed to be mounted directly to the exposed, threaded ends of the U-bolt shackles. coupled to two sets of leaf springs (see 112 of Fig. 1). Each leaf spring set is coupled to the beam axle or axle housing (in the case of a live axle) with a pair of U-bolt shackles113, which are tied together beneath the axle or axle housing with a flat tie plate 114 that is secured with four standard nuts 115 (two on each U-bolt). The mounting system is designed to be mounted directly to the exposed, threaded ends of the U-bolt shackles 113.

Although various mechanical means, such as cables and gears, have been used in the past to deploy chain traction systems, the current genre of chain traction systems relies almost exclusively on pneumatic cylinders for deployment. The primary problem associated with chain traction systems deployed by pneumatic cylinders is that the system may be too bulky for certain applications, such as installation on light-duty pickup trucks. One major problem associated with prior art gear-driven deployment systems is that uneven road surfaces imposed a potentially destructive shock load on the gear train when the chain traction system was in a retracted state. The shock loads had a tendency to shear the teeth off of gears in the deployment gear train. The shock loads could also fracture the housing used to contain the gear train. Another major problem associated with gear-driven deployment systems is that of grit, water, and corrosion related to inadequate protection of the gear train. For a gear-driven deployment system to function reliably, it is essential that all gears and all bearings be completely sealed from the harsh environment beneath the vehicle. Without proper sealing, the life expectancy of such systems would likely be no more than one winter season. Gear driven deployment systems for a chain traction system, if not manually operated, require

some type of motor for automatic operation. For most vehicles, the only type of motor that makes sense is an electric motor, as electric power is readily available from the vehicle's storage battery. Although the automotive industry has solved the problems related to operation of electric motors in a harsh environment (e.g. engine starter motors), in the case of an electric-powered chain traction system, there is still the problem of how to start and stop the electric motor at the appropriate times. If limit switches are to be used, they must be completely sealed in order to protect their delicate circuitry.

What is needed is an electric-powered, gear-driven deployment system for chain traction systems that: (1) is sufficiently compact for installation on a wide variety of vehicles; (2) is completely sealed from the environment; (3) is relatively simple to install and operate; (4) is not subject to damage from shock loads imposed by uneven road surfaces; and (5) solves the problems related to limiting the travel of the device during deployment and retraction.

SUMMARY OF THE INVENTION

The present invention answers the needs expressed in the foregoing section. A new electric-motor-powered, gear-driven deployment system for rapidly-deployable chain traction systems is provided that is compact, sealed from the environment, relatively simple to install and operate, protected from shock load damage to the gear train, and utilizes a timed deployment sequence to avoid the use of limit switches in the harsh environment beneath the vehicle. The deployment system for each vehicle requires at least a pair of deployment units: one each for right and left wheels.

A single unit of the new deployment system includes a housing to which an electric drive motor is externally mounted. By reversing the polarity of the electric current, the direction of revolution of the electric drive motor can be reversed. The output shaft of the drive motor extends through an input aperture within a wall of the housing. For a preferred embodiment of the invention, the output shaft is fitted with an 18-tooth drive spur pinion gear. The spur pinion gear drives a 72-tooth driven spur gear for a gear reduction ration of 4:1. The driven spur gear is mounted on an intermediate drive shaft that is rotatably mounted within the housing. For a first embodiment of the invention, torque applied to the output shaft is limited by a spring-loaded clutch axially mounted on the intermediate drive shaft that couples the driven

spur gear to the intermediate drive shaft. For a second embodiment of the invention, the spring-loaded clutch is eliminated, altogether, and torque applied to the output shaft by the electric motor is limited by MOSFET H-Bridge circuit which limits current drawn by the electric drive motor to a preset maximum. For either embodiment, the spur gears may be replaced with a more costly helical-gear or bevel-gear train with a similar gear reduction ratio. The intermediate drive shaft incorporates a worm that is spaced from the driven spur gear and spring-loaded clutch. The worm operates on a worm gear that is coupled to an output shaft via a spring-loaded damper. The output shaft extends through a sealed output aperture in the housing. The deployment arm of a rapidly-deployable chain traction system (including the rotatably attached friction drive disc and associated chain segments) is rigidly affixed to the end of the output shaft that is external to the housing. The spring-loaded shock damper protects the gear train against shock loads imposed by rotational moments applied to the output shaft by the lever arm, which are caused primarily by forces attributable to uneven road surfaces acting on the rotationally unbalanced mass of the combined deployment arm, friction drive disc and associated chain segments. The spring constant of the coil spring employed within the shock damper is selected as a function of the moment of inertia of the rotationally unbalanced mass.

In order to deploy the system, current is applied to the electric motor for a preset period of time. The time period is slightly greater than the measured time for full deployment from a fully retracted position. Slippage of the spring-loaded clutch on the intermediate drive shaft ensures that full deployment will occur. In order to retract the system, current of reverse polarity is applied to the electric motor for a preset time period that is slightly greater than the measured time for full retraction from a fully deployed position. As a practical matter, the time required for deployment should equal the time required for retraction, as the speed of the motor is identical in both directions and the amount of output shaft rotation is the same in both directions.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a rear elevational view of a typical modern prior-art rapidly-deployable chain traction system;

Figure 2 is a side elevational view of a typical modern prior-art rapidly-

deployable chain traction system;

Figure 3 is a side elevational view of a rapidly-deployable chain traction system which incorporates the new electric-powered, gear-driven deployment system;

Figure 4 is a top plan, partial cut-away view of one unit of a clutch-equipped first primary embodiment of the new electric-powered, gear-driven deployment system, with the top cover removed to show internal components;

Figure 5 is a front cut-away view of one unit of the first primary embodiment of the new electric-powered, gear-driven deployment system, showing internal components and electrical terminals for motor connection;

Figure 6 is a top plan view of the coil spring used for shock decoupling of the drive shaft from the gear train.

Figure 7 is a perspective view of one unit of the first primary embodiment of the new electric-powered, gear-driven deployment system with the top cover removed to show internal components;

Figure 8 is a perspective view of one unit of the first primary embodiment of the new electric-powered, gear-driven deployment system;

Figure 9 is a top plan view of the first primary embodiment of the new electric-powered, gear-driven deployment system;

Figure 10 is a graph of voltage, rotation of the intermediate drive shaft and rotation of the clutch with respect to the intermediate drive shaft as a function of time during system deployment;

Figure 11 is a graph of voltage, rotation of the intermediate drive shaft and rotation of the clutch with respect to the intermediate drive shaft as a function of time during system retraction;

Figure 12 is a top plan, partial cut-away view of one unit of a clutchless second primary embodiment of the new electric-powered, gear-driven deployment system, with the top cover removed to show internal components;

Figure 13 is a system block diagram of a dual electric drive motor controller having current limiting capability;

Figure 14 is a block diagram for a printed circuit board having a dual motor controller with current limiting capability; and

Figure 15 is a detailed system wiring diagram.

PREFERRED EMBODIMENT OF THE INVENTION

In accordance with the present invention, a new gear-driven deployment system for a rapidly-deployable chain traction system is provided. A single unit of the new gear-driven deployment system will be described with reference to the attached drawing figures. It should be understood that a complete system requires the use of at least two units: one for each drive wheel. The drawing figures are intended for the purpose of illustrating a preferred embodiment of the invention only, and not for purpose of limiting the same.

Referring now to Figure 3, a single unit 300 of the new gear-driven deployment system is shown mounted to a beam axle 101 using the mounting system shown in Figure 2. It will be noted that the primary difference between the deployment system shown in Figure 2 and that shown in Figure 3 is that the pneumatic cylinder 110 of Figure 2 has been replaced by a single gear-driven deployment unit 300 and the mounting bracket 109 of Figure 2 has been replaced with a modified mounting bracket 301, which incorporates an angle bracket extension 302 having a sealed ball-bearing race (not shown) installed therein. The sealed ball-bearing race functions as an end support for the output shaft (see item 417 of Figure 4) of the gear-driven deployment system unit 300. The deployment arm 303 is affixed to the output shaft 417.

Referring now to Figure 4, a single unit of the new deployment system 300 includes a housing 401 to which an electric drive motor 402 is externally mounted. The electric drive motor 402 is enclosed in a water-tight motor canister 403. By reversing the polarity of the electric current applied to the electric drive motor 402, the rotational direction of the electric drive motor 402 can be reversed. The electric drive motor 402 has an armature shaft 404, which extends through an input aperture (not shown) within a side wall 405 of the housing 401. For a preferred embodiment of the invention, the output shaft 404 is fitted with an 18-tooth drive spur pinion gear 406. The spur pinion gear 406 drives a 72-tooth driven spur gear 407 for a gear reduction ration of 4:1. The driven spur gear 407 is mounted on an intermediate drive shaft 408 that is rotatably mounted within the housing 401. For a first primary embodiment of the invention, the driven spur gear 407 rotates on the intermediate drive shaft 408, and is coupled thereto via a spring-loaded clutch 409, which has a biasing spring 410 and a driven disc 411 that is secured to the intermediate drive shaft with a first roll pin 412. The compressed biasing spring 410 is sandwiched

between a retainer washer 413 that is held in place on the intermediate drive shaft 408 by a second roll pin 414, and the driven spur gear 407. A paper friction disc 415 is placed between the driven spur gear 407 and the driven disc 411. As an option, the spur gears 406 and 407 may be replaced with a more costly helical-gear or bevel-gear train with a similar gear reduction ratio. The intermediate drive shaft 408 incorporates a worm 416 that is spaced from the driven spur gear 407 and the spring-loaded clutch 409.

Referring now to both Figure 4 and Figure 5, the worm 416 operates on a worm gear 417 that is coupled to an output shaft 501 via a spring-loaded damper 502. The combination of the worm 416 and the worm gear 417 serve to lock the output shaft 501 in any position to which it is rotated. The spring-loaded damper 502 includes upper and lower nested, coaxial, closed-end cylinders (503 and 504, respectively), each of which has an arcuate portion of its cylindrical wall removed, and which, together, form a can in which is housed a coiled, normally-unloaded shock-absorbing spring 505. The edges of the remaining cylindrical walls 506R, 506L, 507R and 507L act on the ends of the shock-absorbing spring 505. Rotational moments applied to the output shaft 501 in either direction cause circumferential loading, as opposed to torsional loading, of the spring. The output shaft 501 extends through a sealed output aperture (not shown in this view) in the housing 401. The deployment arm 303 of a rapidly-deployable chain traction system is rigidly affixed to the end of the output shaft that is external to the housing. The friction drive disc 105 is rotatably attached to the deployment arm 303 and a plurality of chain segments 106A, 106B and 106C are attached to the friction drive disc 105. The spring-loaded shock damper 502 protects the gear train against shock loads imposed by rotational moments applied to the output shaft 501 by the deployment arm 303, which are caused primarily by forces attributable to uneven road surfaces acting on the rotationally unbalanced mass of the combined deployment arm 303, friction drive disc 105 and associated chain segments. The spring constant of the shock-absorbing spring 505 (see Fig. 6) employed within the shock damper 502 is selected as a function of the moment of inertia of the rotationally unbalanced mass. It will be noted that a pair of electrical terminals 507U and 507L are visible in this view. At least one of the terminals is insulated from the housing 401. Electrical connections are made from these terminals 507U and 507L to the electric drive motor 402 through the inside of the housing 401.

Referring now to Figure 6, the shock-absorbing spring 505 is seen more completely in this view. It will be noted that the spring is circumferentially loaded by either spreading the ends 601A and 601B apart or squeezing them together.

Referring now to the isometric view of Figure 7, the housing 401 of a single unit of the new deployment system 300 includes a top cover (see item 801 of Figure 8) a perimetric wall portion 701 and a base 702. The perimetric wall portion 701 may be formed as a single aluminum extrusion. The top cover 801 and the base 702 may be formed from a structural metal such as steel or aluminum. In this view, the shock damper 502 is visible in perspective. The output shaft 501 is axially welded to the upper closed-end cylinders 503. The shock-absorbing spring 505 is also visible in this view. Mounting holes 703 are provided in the base 702.

Referring now to Figure 8, the top cover 801 has been bolted on the housing 401 of the deployment system unit 300 shown in Figure 7.

Referring now to Figure 9, the top view of the deployment system unit 300 shows both mounting holes 703 and the top electrical terminal 507U. An access plug 901 may be removed to expose the end of the intermediate drive shaft 408, which may be equipped with a hex or splined socket which may be engaged with an appropriate wrench in order to raise or lower the traction system in the event of electrical failure.

In order to deploy the system, current is applied to the electric motor for a preset period of time. The time period is slightly greater than the measured time for full deployment from a fully retracted position. Slippage of the spring-loaded clutch on the intermediate drive shaft ensures that full deployment will occur. In order to retract the system, current of reverse polarity is applied to the electric motor for a preset time period that is slightly greater than the measured time for full retraction from a fully deployed position. As a practical matter, the time required for retraction is slightly greater than the time required for deployment, as gravity is working against the system during retraction and the motor power output is identical in both directions. The variation in times required for retraction and deployment may be made identical, or nearly so, by counterbalancing the deployment arm 303.

Referring now to Figure 10, graph A is representative of the voltage applied to the electric drive motor 402, graph B is representative of the rate of rotation of the intermediate drive shaft 408 in radians per second, and graph C is representative of the rate of rotation of the clutch 409 with respect to the intermediate drive shaft 408

in radians per second, all as a function of time during system deployment. Time t = 0 represents the instant that electrical power of positive 12-14 volts is applied to the terminals of the drive motor 402. Time t = x represents the time when the friction disc 105 contacts the associated tire of the vehicle. At this instant, the intermediate shaft 408 stops turning and the clutch 409 begins slipping. The slippage time may be set to be long enough to compensate for wear of the system over its useful life, which may result in slower motor speed and increased friction among the mechanical components of the unit 300.

Referring now to Figure 11, graph A is representative of the voltage applied to the electric drive motor 402, graph B is representative of the rate of rotation of the intermediate drive shaft 408 in radians per second, and graph C is representative of the rate of rotation of the clutch 409 with respect to the intermediate drive shaft 408 in radians per second, all as a function of time during system retraction. Time t = 0 represents the instant that electrical power of negative 12-14 volts is applied to the terminals of the drive motor 402. Time t = x represents the time when the deployment arm 303 reaches its limit stop. At this instant, the intermediate shaft 408 stops turning and the clutch 409 begins slipping. The slippage time may be set to be long enough to compensate for wear of the system over its useful life, which may result in slower motor speed and increased friction among the mechanical components of the unit 300.

Referring now to Figure 12, a second primary embodiment 1200 of the invention is similar to the first primary embodiment 300 shown in Figure 4, with the exception that the clutch has been entirely eliminated and the driven spur gear 407 to be secured to the intermediate drive shaft 408. The elimination of the clutch permits the housing 1201 to be somewhat shorter than the housing 401 of the first primary embodiment shown in Figure 4. Rather than limiting the torque applied to the output shaft 501 with a spring-loaded clutch 409, torque is limited by limiting the maximum amount of current that is supplied to the electric drive motor 402. The fact that torque produced by the motor 402 is essentially a linear function of the amount of current drawn by the motor, torque may be limited by limiting the amount of current that is supplied to the motor to a preset value.

The system block diagram of Figure 13 shows the building blocks for the electrical system for the second primary embodiment 1200. The major block is the dual motor controller with current limiting capability.

A preferred embodiment of the dual motor controller with current limiting capability is shown in Figure 14. Ideally, we would like to be able to connect an electric motor directly to a chip that controls the power supplied to it. However, most integrated circuit chips cannot pass enough current or voltage to spin a motor. In addition, motors tend to generate electrical noise in the form of current spikes, and can slam power back into the control circuit when the motor direction or speed is changed. Consequently, specialized circuits have been developed to supply motors with power and to isolate the other ICs from electrical problems associated with the motor. A widely used circuit for driving DC motors (whether direct drive or geared) is called an H-bridge. It is given that name because it looks like the capital letter "H" on classic schematics. With an H-bridge circuit, the motor may be driven forward or backward at any speed, even using a completely independent power source. The primary features of a preferred H-bridge circuit are:

• Almost all voltage is delivered to the motor by MOSFET transistors;

• Schottky diodes are used to protect against overvoltage or undervoltage from the motor;

• TTL/CMOS compatible driver chips are used to protect the logic chips, isolate electrical noise, and prevent potential short-circuits inherently possible in a discrete H-bridge;

• Capacitors are employed to reduce electrical noise and provide spike protection to the driver chips; and

• Pull-up resistors are employed to prevent unwanted motor movement while the microcontroller powers up or powers down.

Still referring to Figure 14, the current supplied to each drive motor is varied independently by the FET driver, which is controlled via pulse width modulation. The digital control logic is controlled by dual pulse-width-modulated maximum power limiters, which are controlled by dual current sensors (once for each drive motor) in the dual MOSFET H-bridge circuit. For the preferred embodiment of the circuit, each of the two drive motors is controlled by a MOSFET H-bridge, which in turn is driven by an H-bridge FET driver. Each driver is activated by digital control logic that communicates with an analog current limit controller and a microcontroller. The H-bridge circuits are used to limit the current supplied to the electric drive motors (one drive motor for each drive wheel of the vehicle).

Spring loading of the deployment system is necessary due to the resiliency of

the tires. Were it not for the spring loading of the output shaft, the system would not work nearly as well.

The drive motor controller circuits may be programmed so that the circuit remembers whether the chain system is deployed or retracted. A momentary toggle switch is used to send a deployment or retraction signal to the controller circuit. It is designed to require two inputs within a second or so for safety reasons in case the toggle switch is inadvertently bumped.

A primary feature of the system is that no electrical components, other than cables, are subjected to salt and slush. Only two wires connect to the drive motor. It will be noted that retraction may take longer than deployment, as the chains may need to be pulled from beneath the vehicle's tires. in the slush: only two wires going to the device. By monitoring current, the circuit can tell where the mechanism is in the deployment process.

Referring now to Figure 15, the detailed system wiring diagram shows how the current limiting circuits are integrated into a vehicle's wiring system.

Though only a single embodiment of the invention has been disclosed and described herein, it will be obvious to those of ordinary skill in the art that modifications and changes may be made thereto without departing from the spirit and scope of the invention as hereinafter claimed.